THE INSTITUTION OF PRODUCTION ENGINEERS

JOURNAL

(January, 947, Vol. XXVI, No. 1.)

Contents:

PUBLISHED BY THE INSTITUTION
36, PORTMAN SQUARE, LONDON, W.1

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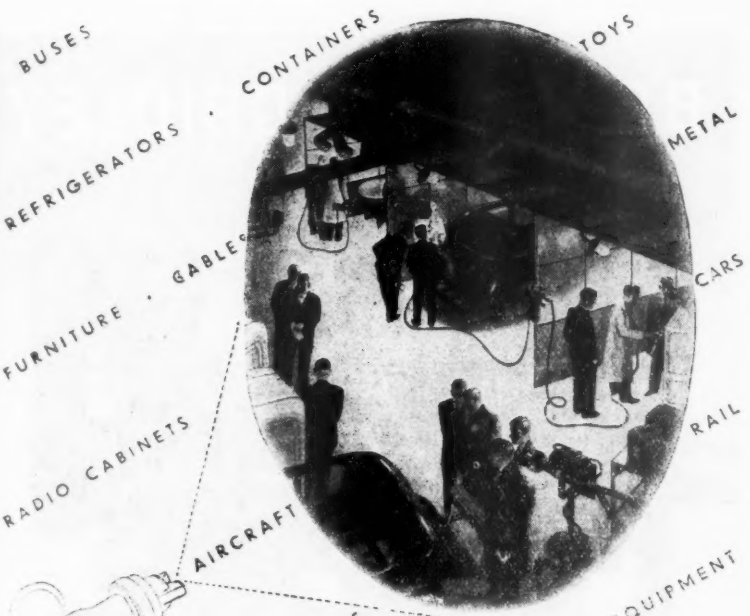

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INSTITUTION NOTES

January, 1947

January Meetings

2nd London Section. A lecture will be given by G. Chelioti, Esq., on "The Structure of Management," at the Institution of Mechanical Engineers, Storey's Gate, St. James's Park, London, S.W.1., at 6-30 p.m.

6th Yorkshire Section. A lecture will be given by Peter Smith, Esq., on "Training Within Industry," at the Hotel Metropole, Leeds, at 7-00 p.m.

8th Luton and District Section. A lecture will be given by H. Eckersley, Esq., A.M.I.Mech.E., M.I.P.E., on "Manufacture and Application of Sintered Carbides," at the Central Library, Luton, at 7-00 p.m.

8th Nottingham Section. A lecture will be given by B. P. Cooper, Esq., M.I.Mech.E., on "Apprenticeship Training," at the Victoria Station Hotel, Nottingham, at 7-00 p.m.

10th North-Eastern Graduate Section. A lecture will be given by T. Burton, Esq., on "Elementary Principles of Cutting Tool Design," at the Neville Hall Mining Institution, Newcastle-on-Tyne, at 6-30 p.m.

11th Yorkshire Graduate Section. A meeting has been arranged at the Great Northern Hotel, Bradford, at 2-30 p.m., when short papers will be given by Graduates and Students.

13th Halifax Section. A meeting has been arranged at the Technical College, Halifax, at 7-00 p.m., at which a sound film, "Steam," by Messrs. Babcock & Wilcox, Ltd., will be shown.

14th Birmingham Graduate Section. A lecture and film show on "The Manufacture of Optical Glass," will be given by Dr. R. E. Bastick, at the James Watt Memorial Institute, Great Charles Street, Birmingham, at 7-15 p.m.

15th Birmingham Section. A lecture will be given by W. C. Puckey, Esq., M.I.P.E., on "Management in Action," at the James Watt Memorial Institute, Great Charles Street, Birmingham, at 7-00 p.m.

January Meetings—*Cont.*

15th Sheffield Section. A lecture will be given on "Research in Relation to Production Engineering" (lecturer's name not yet available), at the Royal Victoria Hotel, Sheffield, at 6-30 p.m.

16th Leicester and District Section. A lecture will be given on "Line Production of Footwear," by E. Eatough, Esq., at the Leicester College of Technology, Leicester, at 7-00 p.m.

16th Glasgow Section. A lecture will be given on "Measurement of Surface Finish," by C. Timms, Esq., M.Sc., A.M.I.Mech.E., at the Institution of Engineers and Shipbuilders in Scotland, 39, Elmbank Crescent, Glasgow, C.2., at 7-30 p.m.

17th Western Section. A lecture, illustrated by slides, will be given by G. H. Asbridge, Esq., on "Crushing Wheels for Form Grinding," at the Grand Hotel, Broad Street, Bristol, 1., at 6-45 p.m.

20th North-Eastern Section. A meeting has been arranged at the Neville Hall Mining Institution, Newcastle-on-Tyne, at 6-30 p.m., at which the following papers will be read: "Profile Machining" by N. J. Cooke, Esq., A.M.I.P.E., and "Preventive Maintenance of Machine Tools," by F. Ward, Esq., A.M.I.Mech.E.

20th Derby Sub-Section. A lecture will be given by W. Whitworth-Taylor, Esq., on "Hydraulics as Applied to Machine Tools," at the Art School, Green Lane, Derby, at 6-45 p.m.

22nd Preston Section. A lecture will be given by V. W. Pilkington, Esq., M.B.E., M.Eng., M.I.Mech.E., M.I.A.E., on "The Training of Apprentices," at the Harris Institute, Avenham, Preston, at 7-15 p.m.

22nd Manchester Section. A lecture will be given by H. E. Chastney, Esq., H.M. Chief Inspector of Factories, on "Factory Inspection," at the College of Technology, Manchester, at 7-15 p.m.

22nd Birmingham Graduate Section. A visit has been arranged to Messrs. Chance Bros. Ltd. Glassworks.

23rd Halifax Graduate Section. A lecture will be given by H. W. Fairbairn, Esq., on "Recent Developments in Die Castings," at the Technical College, Huddersfield, at 7-00 p.m.

January Meetings—*Cont.*

28th Manchester Section. A lecture will be given on "Factory Inspection", by H. A. Hepburn, Esq,, H.M. Inspector of Factories, at Liverpool University, Brownlow Hill, Liverpool, at 7-15 p.m.

31st Eastern Counties Section. A lecture will be given by T. P. N. Burness, Esq., M.I.P.E., on " Production Methods as Applied to Machine Tools," in the Lecture Hall, Electric House, Ipswich, at 7-15 p.m.

31st Lincoln Sub-Section. A lecture will be given by E. R. Walter, Esq., Ph.D., M.Sc., on " Education and the Apprentice," at the Technical College, Lincoln, at 6-30 p.m.

31st Manchester Section. A lecture will be given by H. A. Hepburn, Esq., H.M. Inspector of Factories, on "Factory Inspection", at the Mechanics Institute, Crewe, at 7-15 p.m.

February Meetings

3rd Yorkshire Section. A lecture on production engineering research will be given by Dr. D. F. Galloway, B.Sc.(Hons.)., A.M.I.Mech.E., A.M.I.E.E., M.I.P.E., in the Assembly Hall, Grammar School, Keighley, at 7-00 p.m.

3rd Coventry Graduate Section. A discussion meeting on " The Ideal Training for a Production Engineer."

7th Leicester and District Section. A joint Meeting has been arranged with the Leicester Association of Engineers, to take place at the College of Technology, Leicester, at 7-00 p.m., when the following sound films will be shown: (1) " The Production of Pluto " ; (2) " Through the Mill."

11th North-Eastern Graduate Section. A lecture will be given by J. M. Thirwell, Esq., Int.A.M.I.P.E., on " Production Planning and Control," at the Neville Hall Mining Institution, Newcastle-on-Tyne, at 6-30 p.m.

12th Luton and District Section. A lecture will be given by C. Timms, Esq., M.Sc., A.M.I.Mech.E., on " Measurement of Surface Finish," at the Central Library, Luton, at 7-00 p.m,

12th Manchester Section. A visit has been arranged to Messrs. Ferranti, Ltd., Moston Works.

February Meetings—*Cont.*

13th London Section. A lecture will be given by C. J. Whitcombe, Esq., on "Cold Forging," at the Institution of Mechanical Engineers, Storey's Gate, St. James's Park, S.W.1., at 6-30 p.m.

15th Yorkshire Graduate Section. A lecture will be given by R. Cooper, Esq., A.C.C.A., A.I.I.A., on "Cost Control as an Aid to Management," at the Great Northern Hotel, Bradford, at 2-30 p.m.

17th Derby Sub-Section. A lecture will be given by Miss Shaw, M.A., M.I.P.E., on "Motion Study," at the Art School, Green Lane, Derby, at 6-45 p.m.

17th Halifax Section. A lecture will be given by Dr. J. W. Jenkin, Director of Research, Tube Investments Ltd., on "The Manufacture of Seamless Steel Tubes," at Whiteley's Cafe, Westgate, Huddersfield, at 7-00 p.m.

19th North-Eastern Section. There will be a Discussion Night on "Apprentice Training," opened by W. A. Hildrew, B.Sc.(Eng.)., A.M.I.Mech.E., M.I.P.E., M.I.Mar.E., and G. M. Baker, B.Sc., A.M.I.Mech.E., at the Neville Hall Mining Institution, Newcastle-on-Tyne, at 6-30 p.m.

19th Preston Section. A lecture will be given by A. C. McDonald, Esq., B.Sc., A.R.I.C., M.I.A.E., on "Modern Heat Treatment," at Messrs. Clayton Goodfellow & Co., Ltd., Blackburn, at 7-15 p.m.

19th Sheffield Section. A lecture will be given by Messrs. J. F. Waight and J. Palser, on "Furnaceless Heating by Town's Gas," at the Royal Victoria Hotel, Sheffield, at 6.30 p.m.

19th Birmingham Section. A lecture will be given by Lieut. (E) R. T. M. Toyn, R.N.V.R., M.I.P.E., A.M.I.Mech.E., M.I.B.F., on "Apprentice Training," at the James Watt Memorial Institute, Great Charles Street, Birmingham, at 7-00 p.m.

20th Glasgow Section. A lecture will be given by H. G. Houghton, Esq., on "The Application of Production Standards in Industry," at the Institution of Engineers & Shipbuilders in Scotland, 39, Elmbank Crescent, Glasgow, C.2., at 7-30 p.m.

February Meetings—*Cont.*

20th Leicester and District Section. A lecture will be given by F. G. Rout, Esq., A.M.I.W., on "Fabrication Design as a Substitute for Castings," at a Joint Meeting with the Leicester Association of Engineers, at the College of Technology, Leicester, at 7-00 p.m.

21st Western Section. A lecture, illustrated by slides, will be given by S. Cosgrave, Esq., on "High Frequency Induction Heating," at the Grand Hotel, Broad Street, Bristol, 1., at 6-45 p.m.

21st Manchester Graduate Section. A lecture will be given by L. P. Coombes, Esq., M.I.P.E., M.I.Mech.E., on "Motor Car Production."

22nd Birmingham Graduate Section. An afternoon visit has been arranged to Birmingham Industrial Research Laboratories.

26th Manchester Section. A lecture will be given by W. Symes, Esq., on "Function and Operation of Works Committees in the Engineering Industry," at the College of Technology, Manchester, at 7-15 p.m.

28th Halifax Graduate Section. A lecture will be given on "Powder Metallurgy," at the Technical College, Huddersfield, at 7-00 p.m.

28th Eastern Counties Section. There will be a discussion on "Relationship between Research and Production Engineering," at the Lecture Hall, Electric House, Ipswich, at 7-15 p.m.

Council Meeting

The next meeting of the Council will be held at 11-00 a.m., on 24th January, 1947, at the Institution of Civil Engineers, Great George Street, London, S.W.1.

GRADUATESHIP EXAMINATION, 1947.

1. The Graduateship Examination of the Institution of Production Engineers will be held on Thursday and Friday, April 17th and 18th, 1947.

2. Forms of Application are obtainable either from the Head Office of the Institution or from the Honorary Secretary of any Section of the Institution, and must be despatched *so as to reach the*

Head Office not later than 31st January, 1947, and must be accompanied by an entrance fee of ten shillings.

3. Candidates must be under 28 years of age.

4. Rules and Syllabus and past Examination Papers, price 3d., may be obtained from Head Office.

5. Full details of the Examination will be sent to every candidate.

Personal

Mr. W. G. Thorpe, M.I.P.E., F.I.F.M., has been elected Managing Director of Aircraft General Supplies Ltd., Crescent Works, Erleigh Road, Reading, Manufacturers of Automatic Repetition Components.

Mr. W. Gwinnett, A.M.I.P.E., has now left this country for Australia, where he is taking up a permanent appointment as Director and Works Manager of Rubery Owen and Kemsley (Pty.) Ltd., (a subsidiary of Rubery Owen & Co., Ltd., Darlaston), at Finsbury, near Adelaide, S. Australia.

Books Received

Forming of Aluminium and Its Alloys by the Drop Stamp. Published by the Aluminium Development Association. Price 1/- net.

Issue of Journal to New Members

Owing to the fact that output has to be adjusted to meet requirements, and in order to avoid carrying heavy stocks, it has been decided that the Journal will only be issued to new Members from the date they join the Institution.

Important

In order that the Journal may be despatched on time, it is essential that copy should reach the Head Office of the Institution not later than 40 days prior to the date of issue, which is the first of each month.

IRON CASTING SUPPLIES

The lack of adequate supplies of Iron Castings to meet the demands of increasing commercial manufacturing programmes is causing much worry to Production Engineers, who are experiencing difficulty in balancing production programmes as they are unable to place reliance on regular supplies of good iron castings. They are therefore impelled, in collaboration with the designer, to consider the introduction of expedients which in the main do not fully meet the case.

Much is being done by member firms of Ironfounder Associations to alleviate the labour shortage situation by introducing more mechanisation and at the same time overhauling foundry conditions which, without doubt, have in many cases lagged behind other branches of the engineering industry.

The problem is a national one and not easily solved, but deserves to be tackled with much concentrated thought and enthusiasm. We have received a letter on this subject from Sir Alfred Herbert, K.B.E., Past President of the Institution of Production Engineers, who is eminently qualified, with his broad vision and long experience of meeting difficult situations, to analyse the present problem and put forward sound observations. His views are published below, and it is felt that members will find them of great interest.

"THE ART AND MYSTERY OF THE FOUNDRYMAN"

By Sir ALFRED HERBERT, K.B.E.

Munitions of war make little demand on the iron foundry. There are no iron castings in guns, rifles, ammunition or bombs, and only a very small proportion in aeroplanes, tanks and naval equipment. Very naturally, therefore, our foundries were neglected in war-time, and many foundrymen left their jobs under the attraction of other branches of engineering where, after a very short time, higher wages could be obtained on simple repetition jobs. Apprentices who would normally have taken up foundry work were similarly diverted.

Now we have returned to the arts of peace, and here the demand for iron castings is proportionately enormously higher. In a machine tool, for instance, at least 75% of the total weight consists of cast iron, and in many other branches of engineering the percentage approaches this figure.

We engineers, therefore, find ourselves confronted by a most unfortunate and unprecedented set of circumstances, and unless we can find some relief our output must remain limited. On the one hand, the demand for castings (and good ones at that) is greater than ever before, and the sources of supply very much less. By and large,

our foundries are as numerous and their capacity as great as before the war, and the whole difficulty is that adequate labour does not exist.

Besides the drain on foundry labour during the war, we have to realise that certain psychological factors are at work which are discouraging moulders—who have left the trade—from returning to their jobs, and are similarly discouraging apprentices and young labour from entering the foundry trade.

We hear a good deal about snobbery, which is generally laid at the door of what used to be known as the "privileged classes." Dukes, for instance. Actually the sin of snobbery is far more prevalent among the various grades of workers in industry. We know how often the black-coated worker is liable to fancy himself superior to the manual worker, or how the machine operator or fitter thinks his job more important than that of the moulder, who is supposed to spend his life merely in digging holes in black sand and getting rather dirty in the process, and who is thought to possess less skill and ability than the mechanic.

This is a totally wrong and perverted view, for a moulder who knows his job from end to end possesses a degree of craftsmanship unexcelled by the workers in any other branch of the trade. The moulder has to know his materials—sand and its treatment—the art of core making, and he must know something about metals—their mixtures and treatment—and when he makes a scrap casting (not a very rare event unfortunately) he should know the reason why. He should understand enough about pattern making to be able to criticise his patterns intelligently, and above all he must realise that hot metal has its dangers and that carelessness in its use may involve risk to himself and to his mates.

Modern foundries are moving more and more towards mechanisation and the use of elaborate machines for moulding, sand preparation and for fettling. The moulder's job, it is true, involves harder work and blacker hands than that of the average mechanic, but reasonably hard work hurts no one, and black hands are quickly cured by the use of the washing facilities which are almost invariably available today. Further, black sand washes off more easily than grease and oil.

Much greater attention is being paid to reducing dust and to improving generally the conditions under which foundry work is carried on. It is the invariable experience of apprentices who have transferred—perhaps at first somewhat reluctantly from the machine shop to the foundry—that the work is of absorbing interest.

The proud mechanic must realise also that without an adequate supply of castings he would soon be out of work ; that he depends, in fact, directly on the skill and industry of the moulder for his daily bread.

So much for diagnosis ! Now let us consider what is to be done about it.

In a somewhat desperate effort to meet the difficulty, a good many engineers are substituting castings in aluminium alloys for iron castings. Wherever these can be used for elements which are in motion, the saving of weight is helpful, but in many cases, particularly in machine tool work, weight in the frames, beds and bodies of machines is a distinct advantage for reasons of stability and for reducing vibration.

Another direction in which relief is being sought is in the use of fabricated steel frames instead of castings, but this field is a limited one.

As regards the long term policy, foundry owners must endeavour to improve the position by still further mechanisation and by making their foundries more attractive.

Regarding the short term position, which is indeed becoming desperate, the only solution is to increase the immediate supply of labour, and it must be remembered in this connection that owing to age and death there is inevitably a continuous wastage to be made up.

Serious thought is now being given to obtaining castings from the Continent, where large productive capacity exists, and before very long it is probable that some supplies of castings will be reaching us from this source. It is, however, a remedy which is not without its drawbacks. It means sending money abroad, which is quite contrary to the national policy, instead of spending it at home.

Another solution, and a better one, is the employment in British foundries of existing and available foreign labour—German, Polish or Italian—and this would give great help in our immediate difficulties. But although the Trade Unions have agreed "in principle" to the employment of this labour, there is still great local resistance to it, which is extremely unwise in view of the dependence of all other classes of labour on an adequate supply of castings. Probably these unwarranted objections will be overcome when the situation becomes more clearly realised by the workers. Let us hope this will happen before it is too late.

It has been suggested that the Institution of Production Engineers cannot do much to help, but I disagree entirely with this point of view. By and large, the Institution consists mainly of young men of energy, determination, and ambition, who mean to make careers for themselves in their chosen trade. It is a fact, and an unfortunate one, that a comparatively small proportion of these young men look upon themselves as future executives and administrators in foundry work, and consequently they are not preparing themselves for it.

The higher positions in the foundry industry demand, of course, not only a knowledge of the craft, which can only be gained by actual foundry experience, but at least a working knowledge of

chemistry, metallurgy and combustion, to say nothing of administrative ability and knowledge of human nature.

The supply of young men available to fill these positions is entirely inadequate, but it is certain that for as long as one can reasonably see into the future, there will be abundant opportunity on foundry work for men possessing the necessary qualifications.

The officers and members of the Institution of Production Engineers, if they fully recognize the seriousness of the position, can do much to improve it by arranging for papers to be read and for lectures to be given on foundry work, not only on its technical aspect but on its vital economic importance to the great industry with the progress of which their own careers are so closely bound up.

The Institution can also exercise its great influence on the Directors of Technical Colleges by impressing on them the necessity of providing classes in subjects relating to foundry work, particularly in metallurgy and chemistry of cast iron. The Institution frequently arranges for its members to visit engineering works, where they are always welcome, and it would be a great advantage if visits to foundries were also arranged.

For my own part (and it is over sixty years since I completed my apprenticeship) I have never ceased to be sorry that I had no opportunity of spending any time in the foundry when I was a young man, and I know of a good many other engineers who share my regrets.

SOME OBSERVATIONS ON ROLLING BEARING TECHNIQUE

By R. K. ALLAN, A.M.I.Mech.E., M.I.P.E.

Presented to the Institution of Production Engineers, London Section, November 14th, 1946.

If convincing evidence were required as to the extent and importance of ball and rolling bearings in modern engineering, recollection of the concentrated attention paid to enemy bearing plants by our bombers in recent years would serve the purpose. Contrasting the present position of the industry with that during the 1914-1918 war leaves no doubt about the progress made in the intervening period. During the previous conflict, and for a time afterwards, much ink was spilled and many heated arguments took place about the relative merits of this or that bearing construction, chiefly because the fundamentals were incompletely understood, and practical experience was not sufficiently extensive. A great amount of research work and technical progress since 1918 has eliminated most of the old arguments.

It is not the intention of this paper, therefore, to set before you an array of facts and figures in support of one type of ball or rolling bearing, or against another, for it can be said with some truth that each type now surviving after some 50 years of development has functional characteristics particularly suited to certain combinations of operating conditions. Nor is any attempt made to draw comparisons between these and various kinds of plain bearing, however promising such a theme might be in providing a basis for discussion. Furthermore, details of manufacture are not under review, even though such information is conceivably of special interest to you as Production Engineers.

You will no doubt concede the point that trouble-free bearings are of considerable importance in keeping production machinery on the job, whether it be in a machine-shop, saw-mill, a steel rolling-mill, a textile mill, or a food factory. Conversely, troublesome bearings can be a prime cause of production delays. From this point of view, therefore, reference to certain bearing characteristics of practical significance and to a few recent developments may be of interest.

To begin with, a few general comments will serve as pointers to our main theme:

1. Correct bearing selection is not opposed to commercial considerations. Undue emphasis on low initial cost can often be very expensive in relation to machine life and productive efficiency.

2. Haphazard bearing selection, scanty attention to machine details, workmanship below a certain standard, and careless maintenance, are sure forerunners of unsatisfactory operation and

premature failure. In these circumstances, the bearing and its maker often suffer unjust criticism.

3. When the initial choice of a bearing is correct, and all other details are in harmony, the cliché " fit and forget " is very apt.

4. Fitters should be made to realise that while rolling bearings are capable of carrying relatively heavy loads, and of rotating at high speeds, they are somewhat sensitive to abuse. This means care in handling and skilful fitting.

5. Fitters should be provided with suitable tools, and precise instructions on the details of assembly.

6. The user of a machine is entitled to expect that its bearings are correctly chosen and fitted, and not merely introduced to provide talking points for salesmen. Close co-operation between the machine designer and bearing technicians in the initial stages of design is fruitful of good results. To that end, both must be fully informed—the former about relevant bearing details, and the latter about operating conditions.

7. Machine users should be given detailed instructions as regards maintenance.

Fundamentals of Bearing Design.

Before proceeding further, certain elementary but fundamental conceptions require attention.

Every rolling bearing basically consists of one or more rows of rolling elements with a pair of tracks on which they roll. That elusively simple combination has been known and employed for centuries, but complications appeared when engineers began to take advantage of its possibilities. The science of rolling motion then became a highly specialised subject.

This paper deals particularly with rotary motion, and Fig. 1 shows a plane of rolling in the simplest possible type of radial bearing. The rolling elements and tracks must be circular in every radial plane, and in the case illustrated the tracks are formed directly on the shaft and in the housing. Since it is inconvenient and impractical to make the shaft and housing of the hardened high-grade steel necessary, separate rings are usually employed. The rolling elements all rotate about the bearing axis in the same direction as the shaft, but at a slower speed, and it is seen that where they touch each other the rubbing speed is twice their peripheral speed since at that point they are moving in opposite directions.

Fig. 2 gives a simplified picture of rolling bearing geometry and development. From these diagrams it is clear that, within limits, the tracks and rolling elements in cross-section may be any shape we choose, which accounts for the great variety of designs on which inventors have pinned their hopes, and on many shed their tears.

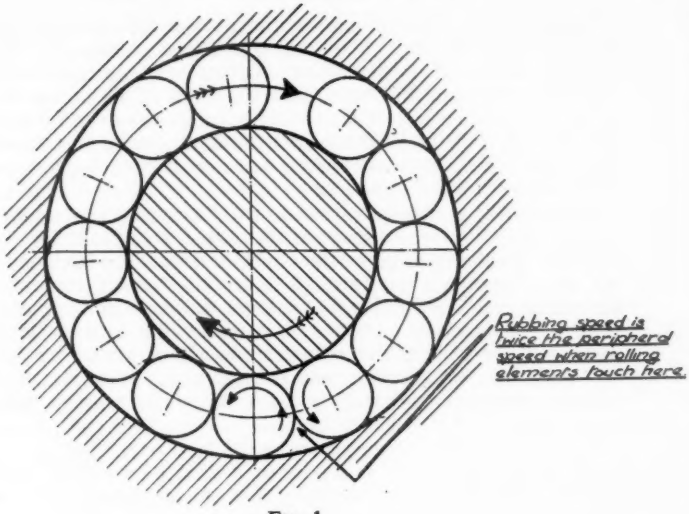

Rubbing speed is twice the peripheral speed when rolling elements touch here.

FIG. 1

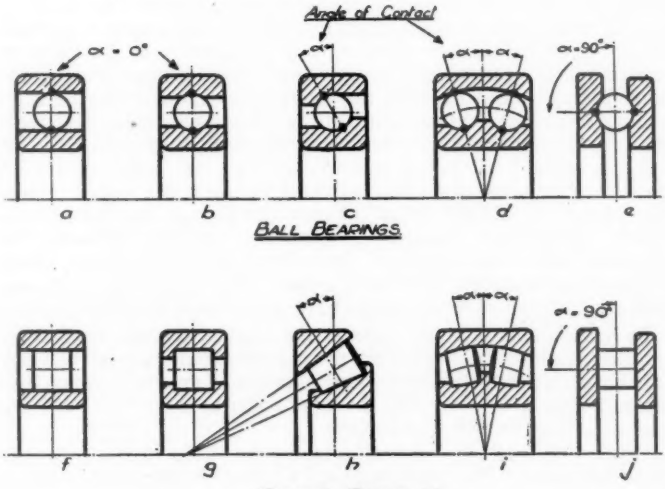

Angle of Contact
α = 0°
α
α
α = 90°
a
b
c
d
e
BALL BEARINGS
α
α = 90°
f
g
h
i
j
ROLLER BEARINGS

FIG. 2

Diagrams "a" to "e" are sectional views of some typical ball bearings, and "f" to "j" of rolling bearings. Obviously, the ball bearings obtain their distinctive characteristics entirely from the sectional profiles of the tracks, since the shape of the rolling element is the same in each case. Each rolling bearing shown is the functional counterpart of the ball bearing immediately above it. It is thus made clear that the differences between ball and rolling bearings are of detail rather than of principle.

From these geometrical variations two features of vital importance emerge, namely, size and shape of the contact areas, and the angle of contact—usually denoted by the Greek letter α. The letter is simply the angle between a line through the rolling element contacts with the tracks, and a plane at right angles to the axis. In diagrams "c" and "h", the contact angle may be varied within fairly wide limits, and in "d" and "i" within narrow limits ; in diagrams " a", "b", "f" and "g" it is 0°, sometimes referred to as " straight contact," while in diagrams "i" and "j" it is 90°, as in bearings for pure thrust load.

Types "a" and "f" can carry radial load only, that is, at right angles to the axis. "a" is a poor load carrier because of its small point-contact areas; "f" is a good load carrier because of its larger line-contact areas.

Type "b", the normal single-row rigid type, has grooved tracks which give increased size of the ball contact areas and also enable it to take axial load in either direction. When axial load is applied, the rings shift axially in relation to each other, and the angle of contact increases from zero to some angle which is dependent on the magnitude of the axial load, the internal slackness of the bearing, and the cross-sectional curvature of the tracks. Owing to this lateral shift this bearing cannot, by itself, position a shaft very closely in an axial direction. Its roller counterpart "g" can take axial load because of the side flanges on both rings, but only to a limited extent, since there is sliding contact between the flanges and roller ends which necessitates efficient lubrication. Greatest efficiency is obtained when the length of the rollers does not exceed about $1\frac{1}{2}$ times the diameter. With long rollers, efficient guidance becomes difficult. Needle roller bearings have long, small-diameter rollers.

By varying the contact angle in the angular-contact type "c", various radial to axial load-ratios are obtained, but the axial load can only be taken in one direction owing to the small groove depth on one side of the tracks. Obviously, the taper-roller bearing "h" can only take axial load in one direction.

Types "d" and "i" are self-aligning by virtue of the spherical outer ring track. The roller bearing "i" has a higher load capacity than any other of the same dimensions. In both types the load is evenly distributed automatically.

Thrust bearing "e" has a limited speed range because of centrifugal and precessional effects. For accurate axial positioning of a shaft, however, it is very well suited. Bearing "j" has very limited uses because the inner and outer ends of each roller try to rotate at distinctly different speeds.

Cages, or retainers, as they are sometimes called, are generally necessary to prevent sliding contact between the rolling elements and to keep them from dropping out when the bearing rings are separable. They are made in a great many ways, each kind having claims to some functional perfection or other, or to economy in manufacture.

Many other constructional variations less fundamental in their effects are employed, such as :

slight convexity of one of the tracks in types "f", "g" and "h" to avoid edge-loading of the rollers and to reduce skewing ;

making the large ends of the rollers spherical in types "h" and "i" to secure maximum contact area at the flange ;

making the inner-ring bore taper for use with a tapered journal, or tapered sleeve, to facilitate mounting ; and so on.

Contact Areas.

Size, shape and position of the contact areas in rolling bearings have a profound effect on friction, temperature, load capacity, life, and radial and axial movement under load.

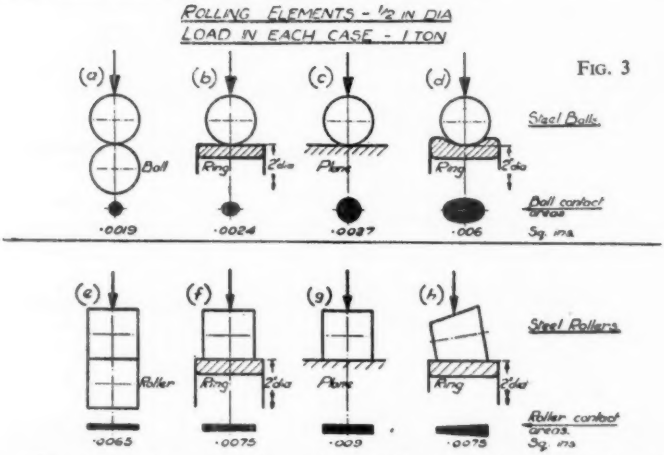


FIG. 3

The actual shape and dimensions of the contact area can be calculated if we know the modulus of elasticity of the material, the shape and size of the contacting bodies, and the load imposed. Fig. 3 shows the form and size of contact areas for rolling elements of

½" diameter contacting with various surface profiles. With a ball the area is either circular or elliptical, with a cylindrical roller, rectangular, and with a taper roller it is wedge-shaped. It is clear that the more closely the supporting surface envelops the rolling elements the greater is the contact area, but this cannot be carried too far, or friction becomes excessive.

For a particular ball bearing with its inner ring shaped as in diagram "d", ten balls of ½" diameter, and an applied radial load of 7,700 lbs., which is the rated capacity for a speed of 15 r.p.m., the maximum ball load would be 3,850 lbs., and the contact area only ·008 sq. ins. This means that the average compressive stress in the area is about 200 tons per sq. in. As a rule, bearings are not loaded to the full rated capacity and this, together with the fact that capacity is reduced as speed increases ensures better stress conditions generally than in the example quoted. The bearing taken as an example probably would not be subjected to more than about 500 lbs. radial load at 500 r.p.m. With that loading, the maximum ball load is 250 lbs. the contact area ·0013 sq. ins., the compressive stress in the area, 90 tons per sq. in., and the radial shift of the inner ring towards the outer ring about ·001".

The compressive stresses just mentioned may seem surprisingly large, but as they are purely local, need not give rise to any misgivings.

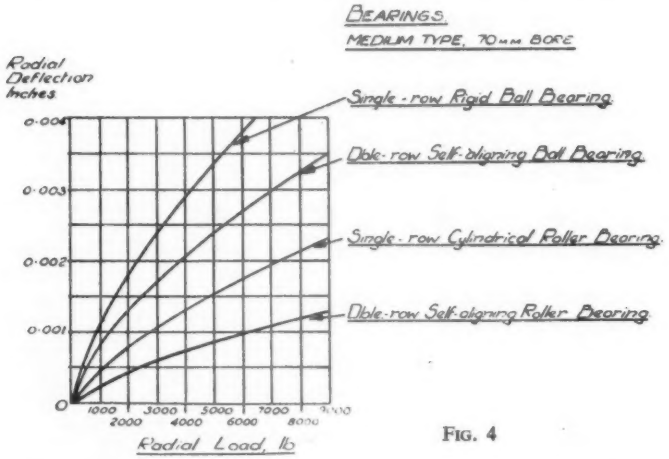


FIG. 4

Proof of that is found in the millions of bearings giving satisfactory service, among which may be mentioned one of 46½" diameter carrying a load of 1,000 tons in a cold-rolling mill.

To those of you who are concerned with precision machine tools, the amount of deformation at the contact areas is of greater interest, and a radial movement of ·001″ possibly considered excessive, particularly if a small amount of slackness within the bearing together with some looseness of fit of the outer ring on its seating has to be added. The remedy here is to make use of the right type and size of bearing for the duty, ensure that the fits are snug enough, and to adopt preloading if necessary.

Fig. 4 gives a graphical picture of the radial deformation of several bearing types subjected to radial load only, from which it is clear that the single row rigid ball bearing yields the most and the double-row self-aligning roller bearing the least. The latter type has been a standard fitting for many years on the work spindles of a range of centre-lathes produced by a well-known maker.

Fig. 5 gives a similar comparison for axial deformation with the bearings subjected to axial load only. The yield of the thrust ball bearings is notably small in comparison with the others. Thrust bearings, however, cannot always be used, because of speed limitations.

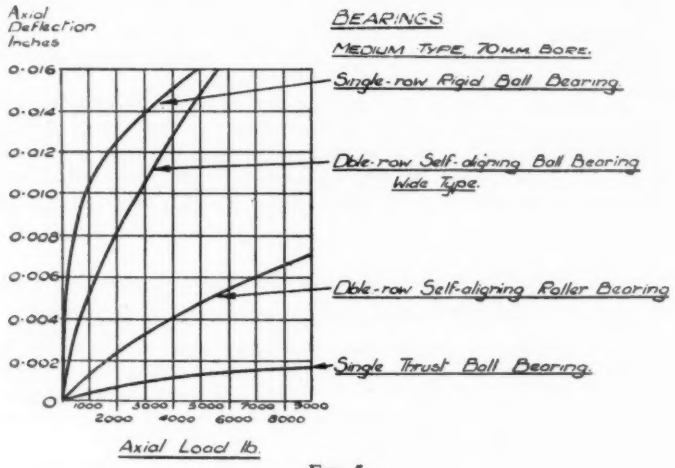


FIG. 5

Preloading.

Preload is a term relating to the condition when a bearing is so adjusted as to subject it to an internal load prior to the application of an external load. The object of the procedure is to reduce the amount of elastic movement that takes place when the external load varies in

magnitude or direction. The method is especially applicable to machine tool spindles that must shift to the minimum extent, either radially or axially, from a fixed position under the influence of and in relation to the cutting tools. Absolute perfection in this respect is impossible with any kind of bearing, but we can at least approach the ideal very closely.

In Figs. 4 and 5, it is apparent that the deflection curves are steep to begin with, and then less steep as the load increases. Preload simply shifts the external load deflection to the flattest part of the curve, and with ball bearings the gain in rigidity is considerable.

Taking an angular-contact ball bearing of $1\frac{3}{4}''$ bore, and with $\frac{1}{2}''$ diameter balls, for example, an axial load of 500 lbs. will cause deflection of about ·0007″. By mounting two of these bearings side by side and adjusting them endwise to give a preload of 500 lbs., then on the application of the external axial load of 500 lbs. the deflection will only be about ·0002″.

With rolling bearings there is less gain by preloading, for the deflection is practically proportional to the load. Rolling bearings, however, clearly yield to an even smaller extent than preloaded radial ball bearings, and preloading them is therefore less essential.

Preloading should not be employed unnecessarily, for it has to be paid for by a reduction of bearing life. The working temperature is also increased, with resultant expansion effects.

An excellent example of preloading and effective bearing selection is given by the lathe live-centre shown in Fig. 6. A thrust ball bearing is used for the axial load, since its axial yield is less than any other type. The taper rolling bearing was chosen for the main radial load because, in addition to small radial yield, it is easily adjusted against

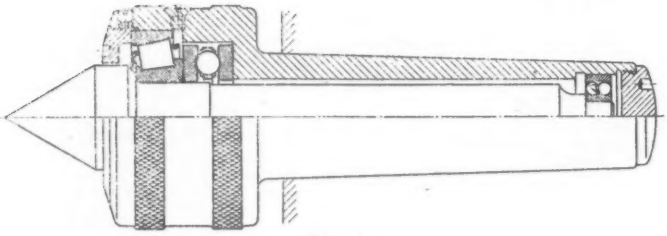


FIG. 6

the thrust bearing for the purpose of eliminating initial radial and axial play. In this case the preload required is not great. The small double-row self-aligning ball bearing acts as a steady, and owing to its distance from the front bearing, any slight radial slackness it may have is negligible.

Plastic Deformation.

Another matter of interest in connection with contact areas is that of plastic deformation. Within the elastic limit of the material, elastic deformation is proportional to the cube root of the rolling body load squared. Owing, however, to minute imperfections on the best of surfaces, even very small loads cause permanent deformation, but this is so small that it can only be detected by a slight change in the reflection of light from the surface.

As the load increases beyond the elastic limit of the material, plastic deformation occurs, that is, the material does not entirely return to its original shape after removal of the load. In bearings rotating under normal loads, the successive indentations blend into each other, the result being a slight change in track contour which is hardly measurable.

In stationary bearings shock load may cause relatively deep indentations—a Brinell effect. Vibration also produces similar indentations known as "false Brinelling," an effect believed to be a form of wear dependent on, or aggravated by the presence of oxygen in the contact areas, and accompanied by fretting corrosion.

True plastic deformation does not cause fracture until it has assumed considerable proportions, and it is only in exceptional cases that it need be considered from this view point. Limitation of plastic deformation to one-ten-thousandths of the rolling-body diameter is now thought good practice for bearings loaded statically and which rotate afterwards, and is ensured by selecting a bearing of sufficient capacity. We therefore have the paradoxical condition that, within limits, a rotating bearing can carry a heavier load than a stationary one. A greater depth of indentation than that just defined can be tolerated in cases where rough running is not objectionable, especially when speed of rotation is low.

Load Distribution Within the Bearing.

A matter that does not always receive the attention it deserves is the proportion of bearing load carried by each rolling element. The diagram and table in Fig. 7 refer to the loading of one only single-row angular contact bearing, either ball or roller type, and convey some idea of what happens. Complete analysis is rather complex, and in fact it was only in 1933 that the problem was solved for bearings with a fixed contact angle. In single-row rigid ball bearings, where the contact angle varies with the axial load, as already mentioned, the conditions are so complex that no generally applicable analysis has as yet been formulated.

Referring to diagram "a", it will be clear that the direction of the load resultant depends on the relative values of W and T. When $T = 0$, the resultant load is in a radial direction and the rings will

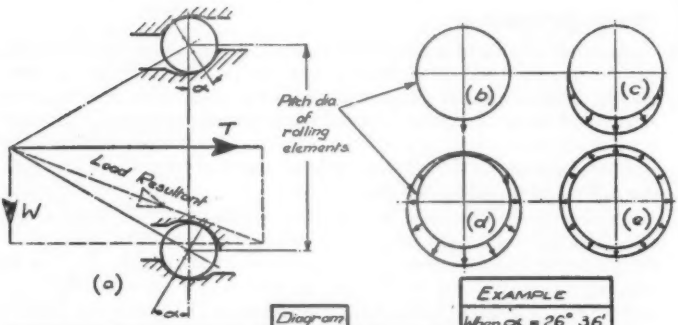


	Diagram		Example When $\alpha = 26° 36'$
Rings move apart	—	$T = 0$	$T = 0$
Load concentrated on one rolling element	b	$\frac{T}{W \tan \alpha} = 1$	$T = \frac{1}{2} W$
Load on half of track	c	$\frac{T}{W \tan \alpha} = 1{\cdot}25$	$T = \frac{5}{8} W$
Load all round except on top rolling element	d	$\frac{T}{W \tan \alpha} = 1{\cdot}7$	$T = \frac{7}{8} W$
Load evenly distributed	e	$W = 0$	$W = 0$

FIG. 7

separate axially unless another force is applied to prevent the movement. When W = 0, the resultant load is in an axial direction, and is distributed equally among the rolling elements as shown in diagram "e". The state of affairs intermediate between these two extremes is shown in diagrams "b", "c" and "d", and the governing conditions indicated in the table. From these details it is seen that the combined-duty capacity varies with the relative values of radial and thrust load.

Diagrams and formulae similar to these can be developed to show the effect of eccentric loading on both single-and double-thrust bearings. With a central axial load on a single-thrust bearing, the rolling elements are obviously equally loaded ; if the load is eccentric to the extent of 0·6 times the pitch radius, one rolling element is unloaded ; if the eccentricity is 0·8 times the pitch radius, only half of the rolling elements carry the load ; and if the eccentricity is equal to the pitch radius all of the load is carried by one rolling element. In a double-thrust bearing, eccentricity of the load line greater than 0·6 times the pitch radius causes part of the load to be transferred from one row of rolling elements to the other, until with a pure turning moment (infinite eccentricity) half of the rolling elements in each row carry the load.

Load distribution within a bearing is of special importance when the bearing does not rotate, since the magnitude of the contact areas is affected, and the indentations do not blend into each other as in a rotating bearing. The life of rotating bearings is also affected by the load distribution, but other factors, including the number of stress cycles at a point on the weakest track, must also be taken into account.

As regards angular-contact bearings, it is obvious that in order to keep the parts of one of these together when radial load is applied, it must be paired with another bearing—not necessarily of the same size and type—and the load reaction of the pair then taken into account. Responsibility for bearing selection based on a correct assessment of individual bearing load lies, of course, with the machine designer. On the other hand, the machine user is not absolved from responsibility if he makes a machine operate under conditions differing greatly from those for which it was originally designed.

Rotational Speed.

The question of speed is no doubt one of special interest to you, for Production Engineers are tremendously keen to make the wheels go faster and faster, and thus get a few more bits and pieces from machines in a given period of time. High-speed bearings help to achieve that end.

Maximum bearing speeds are governed by a number of factors which influence the amount of heat generated, and this in turn sets a limit to revs. per minute. Research on this matter has been carried out by various investigators, but no satisfactory general formula has as yet been developed owing to the difficulty of isolating one factor from another.

Bearing friction-coefficients give part of the answer, for in practice it is found that radial bearings with coefficients of ·001 to ·0015 are best for high speeds.

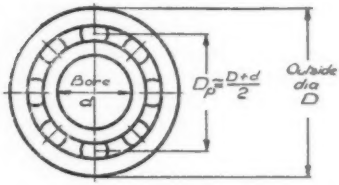
Temperature rises with increase of load, and some kind of cooling arrangement may be necessary in difficult cases.

Lubrication has such a great influence on permissible speed that full advantage must be taken of past experience for guidance in particular cases as to the kind and quantity of lubricant to use, together with the best method of applying it. Fortunately, very small amounts of grease or oil are needed for rolling bearings, and this fact makes the problem less difficult than it might seem.

The cage that spaces the rolling elements also affects the maximum speed attainable to a great extent. Well designed and well made cages pressed from the sheet are very good up to a point, but yellow-metal cages machined from the solid are generally better for high speeds. Cages of reinforced plastic material show considerable

promise, and both the moulded and machined varieties have been used with some success.

Probably by now you are wondering just how, with all these intractable factors, the limit of speed is determined in any particular case. To this natural question there is no well defined answer. In general, however, where lubrication is good and load low, we can be guided by the simple formula and constants given in Fig. 8. Dp is in millimetres.



$$\text{Approx max. r.p.m.} = \frac{C}{Dp \text{ in mm}}$$

Type of Radial Bearing	Type of Cage	C	EXAMPLE d = 25 m.m. D = 52 m.m. Dp = 38·5 m.m. Approx max. r.p.m.
Dble.-row Self Aligning Ball Single-row Rigid Ball Single-row Cylindrical Roller (with short rollers)	Pressed Steel	500,000	13,000
Single-row Rigid Ball	Centred Bronze	700,000	18,000
" " "	Special	1,500,000	40,000
Dble.-row Self Aligning Rolls.	Pressed or Drilled	350,000	9,000
Taper Roller.	Pressed		
Thrust Ball Bearing.	Pressed	200,000	5,000

FIG. 8

Although the table gives a constant for thrust ball bearings, the question of their maximum speed requires fuller consideration. Quite often the matter is dismissed rather airily with the remark that the speed is limited by the centrifugal force of the balls. Actually gyroscopic effects predominate, and the balls endeavour to spin about an axis other than the rolling axis. That spin is opposed by a frictional moment which is proportional to the ball load. In effect, to avoid excessive generation of heat, the ball load must be not less than a certain magnitude for a given speed and bearing dimensions.

Gyratory forces also come into play in radial angular-contact bearings, but their effects are of no real importance.

Limits and Fits.

When two cylindrical parts of a machine have to be paired together, the designer decides how slack or how tight the fit must be. Since absolute accuracy of the mating parts cannot be expected, he must also decide the maximum and mimimum slackness or tightness of fit that can be allowed. This allowance determines the permissible variation or tolerance for the diameter of the mating parts. The tolerance should be great enough to permit economical manufacture, and small enough to ensure that the parts will function as required. In these two respects there can be great divergence of opinion.

External dimensions of rolling bearings are standardised, so that the kind of fit required has to be obtained by varying journal and housing-bore diameters to suit. The effect of the fit on the functioning of the bearing, and the effects of load and temperature on the fit must be kept well in mind.

Radial load tends to loosen the fit of the inner ring on the shaft end of the outer ring in its housing, because of shaft and ring compression; it is therefore clear that heavily loaded bearings in general need tighter fits than lightly loaded ones. If the load changes direction continuously in relation to one of the rings, or vice versa, tight fit is essential to avoid creep, for creep causes wear and induces cracks which quickly bring about ring fracture.

Loose fits have certain advantages in mounting, and when used where the operating conditions cause creep, efforts are sometimes made to stop it by means of grub screws, or keys, or even hard tightening of the shaft nut. These expedients, however, are useless, for apart from the weakening of the rings caused by grooves or keyways, the keys become badly worn, or grub screws shear off; tightening of the shaft nut is ineffective, since an end pressure at least 10 times the radial load is necessary.

As a rule, a loose fit for a bearing ring is admissible only where the resultant load does not shift its direction in relation to the ring, for only then there is no force to cause ring creep.

The spindles of precision machine tools require bearings of special accuracy, but accuracy is lost if the bearing seatings are not truly cylindrical. Seatings that are oval or taper cause the bearing rings to take the same shape, with the result that the axis of the spindle has an eccentric motion additional to any slight out of truth which may be inevitable from the loading conditions.

The red rust that sometimes forms on the contact surfaces of a shaft and the bore of a bearing may have caused some worry on occasion. This condition is not confined to rolling bearings, for it occurs with other fitted parts as well. It is a sure sign that the fit of the parts has not been complete, and that there has been sufficient movement between them to cause fretting. Minute surface particles

dislodged by the movement become oxidised by oxygen present between the surfaces, the resultant ugly patches being known as contact rust, and the action as "fretting corrosion." The condition can take place and spread as a result of local strain under heavy load. In bad cases, the parts become so firmly welded together as to defy attempts to separate them by ordinary methods. The only remedy is to ensure that the fit is tight enough and the surfaces so completely in contact that no relative movement of the surfaces can take place either due to loading or shaft flexure.

Some recent developments will now be referred to.

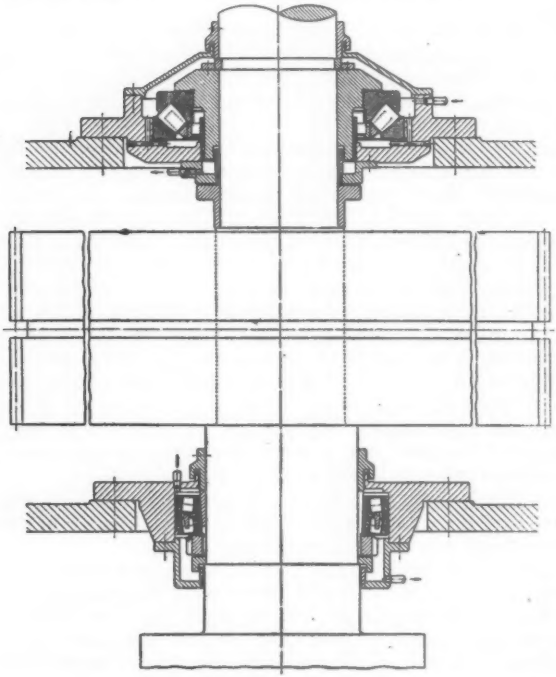


FIG. 9

Spherical Roller Thrust-Bearing.

An interesting design of thrust roller bearing is shown at the top of Fig. 9. The rollers are basically of conical form, but in cross-section the profile is curved in a similar way to those in the double-row spherical roller bearing previously described. Self-alignment is

achieved by the sphered track of the non-rotating washer. The angle of contact is about 45°, which enables the bearing to deal with considerable radial as well as thrust load.

An important feature is the spherical big ends of the rollers, which are so matched to their supporting flange that an oil film is formed between the surfaces during rotation, thus permitting relatively high bearing speeds.

This bearing has a very high load-carrying capacity, and in conjunction with its other characteristics has provided the solution of many rather difficult bearing problems.

Machine Tool Spindles.

For precision work, machine tool spindle bearings must satisfy very exacting conditions, including—

Minimum deflection under varying loads ;

Maximum truth of running under loads of varying magnitude and direction ;

Minimum temperature variation throughout the speed range ;

Adjustability to obtain minimum radial and axial slackness ;

Both rings of radial bearings should be tight on their seatings ;

Bearing sectional height should be small to secure maximum spindle diameter without sacrificing load capacity.

These conditions are not easy to fulfil, and have been made more difficult since the introduction of the cemented carbide cutting tools and negative rake angles.

Fig. 10 shows a lathe spindle bearing scheme in which these

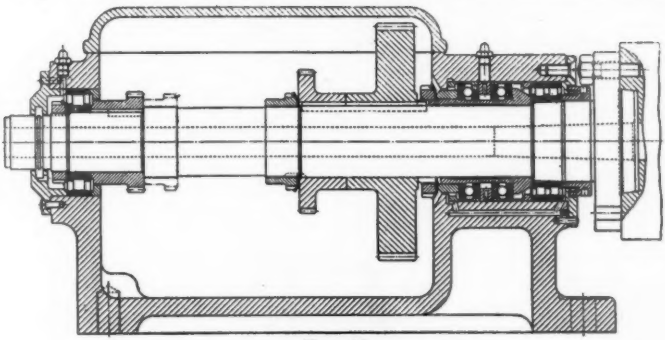


FIG. 10

conditions are met by taking advantage of the various features already discussed. The most important bearing in the design is the new double-row cylindrical type at the work-end, which is the

outcome of extensive experiment and practical tests. Its chief features are low sectional height, a large number of small diameter cylindrical rollers arranged in two rows, a taper bore in the inner ring, and inner-ring tracks ground at one setting while the ring is mounted on a taper mandrel. Elimination of internal slackness is achieved by adjusting the inner ring on the tapered shaft seating.

Next in importance are the ball thrust-bearings which are adjusted practically to eliminate axial movement. The cylindrical roller bearing at the rear end is a standard type with minimum diametric slackness.

Various bearing arrangements embodying the same principles are possible, but with modifications in detail as required. Fig. 11 shows the vertical spindle of a surface grinder, using the two-row cylindrical roller bearing and a single thrust ball bearing at the grinding wheel end. A number of helical springs act upwards on the central single-row ball bearing to keep the thrust-bearing loaded to a predetermined amount according to the speed.

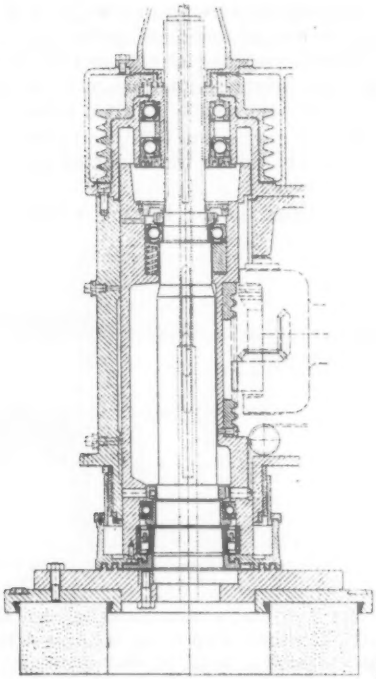


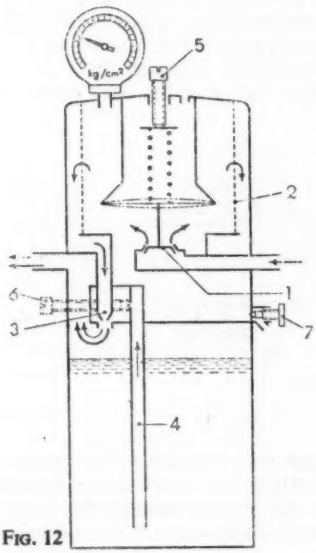
FIG. 11

Oil-Mist Lubrication.

High speed bearings have always presented certain problems to bearing manufacturers, machine designers and maintenance engineers alike. One of the great difficulties is lubrication. Grease lubrication has its limits, while the normal method of oil-lubrication, such as oil-bath, circulatory systems, metered feed, and so on, present various practical obstacles to complete satisfaction.

The solution of the problem appears to have been found in oil-mist lubrication, an idea that is not altogether new. Much experimental work over several years, however, has led to the perfection of the means of producing the oil mist and the methods of applying it.

Fig. 12 shows a diagram illustrating how a patented oil-mist producing apparatus functions. Compressed air enters by the pipe on the right, passes through a reducing valve (1) controlled by screw (5), through a filter (2), and then through the atomising nozzle (3). The vacuum produced in the tube (4) lifts the oil which is then atomised and mixes with the air. Flow of oil is controlled by screw (6). Water of condensation is drained away through opening (7). The glass container is filled with oil through the filling aperture (8).



1. Reduction valve
2. Air filter
3. Atomizer
4. Oil suction tube
5. Air pressure regulating screw
6. Screw for regulating the oil supply
7. Screwed plug closing outlet for condensation water.

FIG. 12

Oil mist produced by this method is quite stable and can be passed for very long distances through suitable piping. The scheme has the following advantages :

Current of air through the bearings effectively prevents entry of grinding dust, or other extraneous matter.

Oil-mist supply can be controlled to a nicety to give exactly the amount of lubricant required by the bearings.

The air currents keep temperature low.

Resistance to rotation of the bearing is less than with other methods of lubrication.

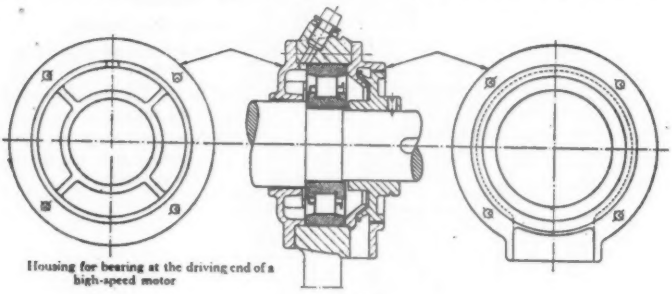
Oil saving is particularly striking. In one place where careful records were kept, oil consumption dropped by 90%. In another case, two double-row self-aligning roller bearings 160-mm. bore on a hot saw at 1,400 r.p.m. only require 2 pints of oil in 400 hours. Normally, oil consumption per bearing is between ·015 to ·06 cu. in. per hour.

Experience indicates that the amount of oil in the escaping air is so minute that even where foodstuffs are being processed, risk of contamination is negligible.

Grease Valve.

If grease lubrication is preferred in cases where the re-lubrication intervals are short and uninterrupted service is essential, a recently-developed grease valve greatly simplifies maintenance.

The arrangement in principle is indicated by Fig. 13. Grease is forced, by means of a grease gun, into the back cover space ; this is



Housing for bearing at the driving end of a high-speed motor

FIG. 13

divided by baffles to prevent mass rotation of the grease. The grease then passes through the bearing, any excess being picked up by the rotating disc on the right and ejected through the large aperture at the bottom of the right hand cover, where it is collected in a suitable

receptacle. By these means, lubrication is easily effected without any danger of packing the housing too tightly with grease.

To be really effective, the device needs careful design.

Easy Fitting and Dismantling by Oil-Injection.

The tight fits so frequently necessary with rolling bearings sometimes make fitting difficult. When the interference is considerable, the direct pressure required to force the parts together may be so great that the expedient of expanding the ring by heating, or contracting the shaft diameter by freezing is resorted to. The same comment also applies to shaft couplings and other parts paired tightly together to prevent relative movement even when subjected to considerable torque.

The difficulties inherent in fits of this class have been overcome in a most ingenious manner by a method applicable to dismantling as well as fitting. Time permits only a short reference to the scheme, which is covered by patents, but the principle will be easily understood as applied to a range of shaft couplings.

One of the smaller couplings is shown in section in Fig. 14. This consists of two parts only, without keyways or bolts, namely an inner sleeve slightly tapered on the outside, and an outer sleeve with

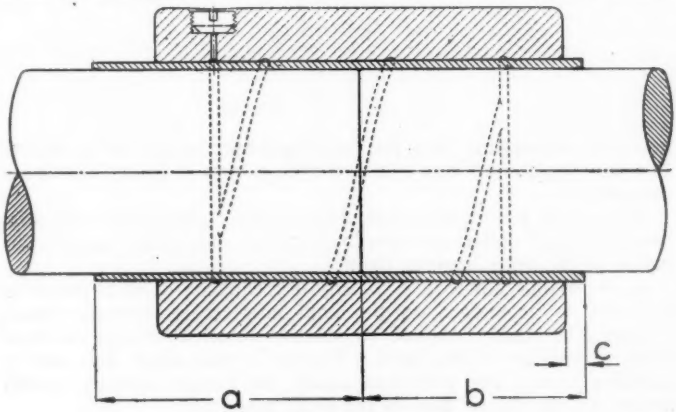


FIG. 14. Small coupling

a slightly tapered bore to match the inner sleeve. The hole at the end of the outer sleeve is tapped to take the end of a simple screw-down type of oil injector, and the inner end of the hole connects with grooves cut on the inside of the outer sleeve.

By forcing oil between the surfaces, the thin inner sleeve is compressed and the outer sleeve expanded, thus making it easy to slide the

outer one along the taper to the required position. On releasing the pressure the outer sleeve contracts, forcing the oil from between the surfaces, thus compressing the inner sleeve to grip the shaft tightly.

Fig. 15 shows one simple type of oil injector. When more oil is required than the injector holds at one filling, a check valve is

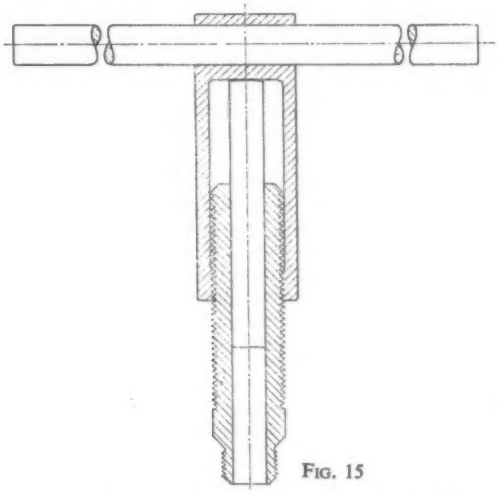


FIG. 15

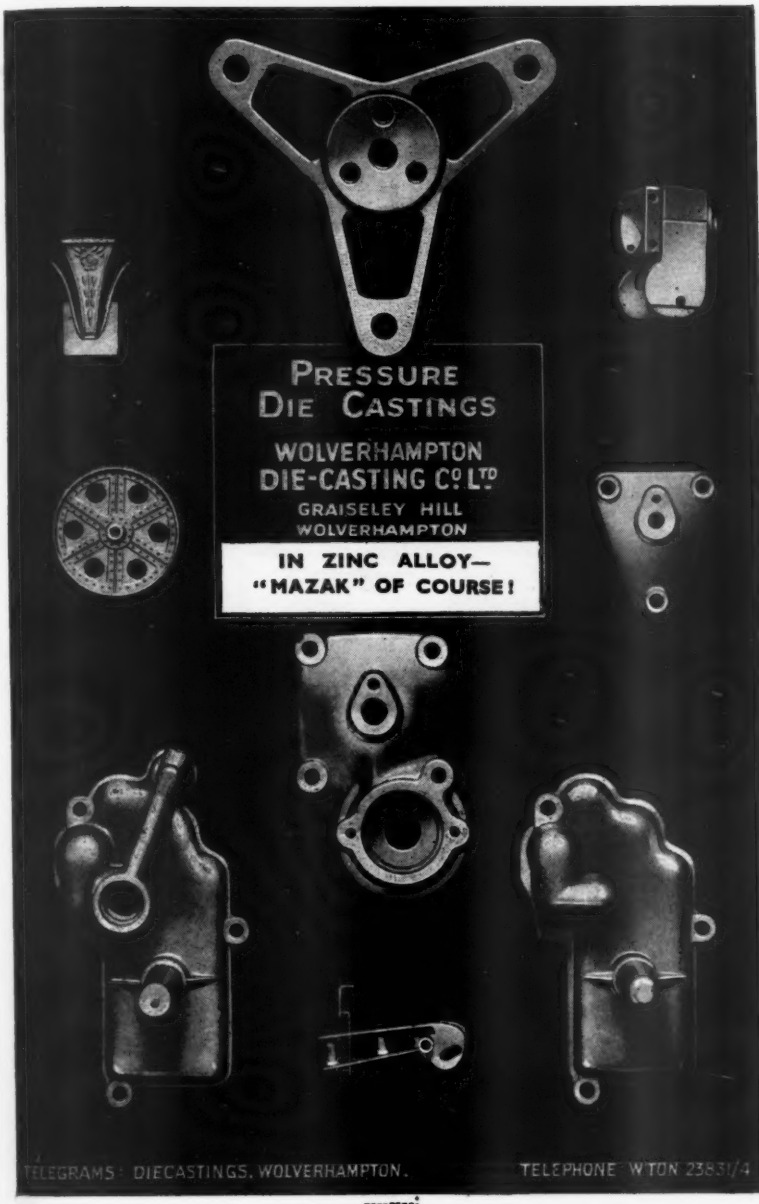
necessary to hold the oil in the coupling while the injector is refilled. For very large couplings, injectors with a large capacity have been designed.

To assist in pulling the outer sleeve of the coupling into position, two U-shaped collars are employed which are pulled together by means of the bolts between them.

As is obvious, the torque transmitting capacity of parts joined in this way is dependent on the final interference between them, a condition readily attained by suitable dimensions and the final relative position of the parts. Extensive tests show that with a coupling joining two mild steel shafts, the torque capacity is only limited by the plastic flow of the shaft material.

Conclusion.

This survey of rolling bearing practice has been necessarily rather condensed and somewhat sketchy, for the scope of the subject is so wide that in the course of a short paper the treatment is bound to be inadequate. Possibly, however, enough has been said to indicate slightly the progress made, the trend of development, and the possibilities.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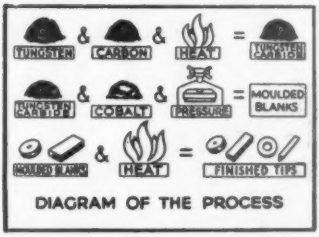

DIAGRAM OF THE PROCESS

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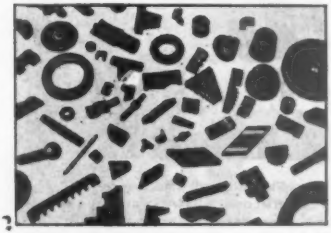

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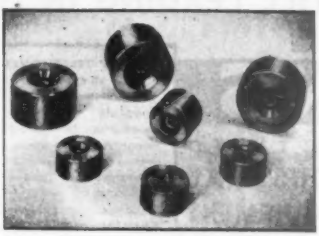

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All communications regarding advertisements should be addressed to the Advertising Managers, T. G. Scott & Son. Ltd., Talbot House, 9, Arundel Street, London, W.C.2. Phone: Temple Bar 1942.
Printed by Maxwell, Love & Co. Ltd. 10-15, Bradley's Buildings, White Lion Street, London, N.1.

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